# STRESSES IN RIVETED JOINTS

BY

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ARMOUR INSTITUTE OF TECHNOLOGY

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Stresses in riveted joints

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# A THESIS

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H. A. R.

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# STRESSES IN RIVETED JOINTS

The stresses on rivets and riveted joints is a subject which is attracting the interest of engineers to a greater extent recently than it has for some time. The technical journals are lately devoting considerable space to the subject. Heretofore the allowed stress on a rivet has been figured from the ultimate strength of a joint and introducing a factor of safety, without making a real scientific investigation of the stresses that actually occur.

It is the purpose of this thesis to determine the actual maximum stresses on a rivet from a theroretical consideration, using only such experimental constants as can not otherwise be obtained.

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PART 1

A consideration of the resistance of the shank to bending moment.

RESISTANCE OF SHANK

TO BENDING MOMENT

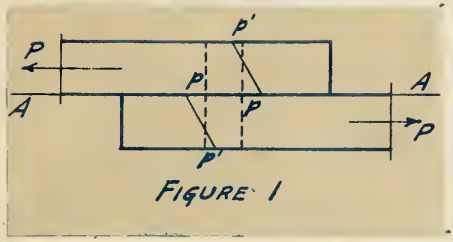



FIGURE 1

Let us consider two plates arranged as in Figure 1 with a rod of unit diameter like the shank of a rivet holding the two from slipping one on another in a lateral direction. A load P is applied to each plate inducing shear in the section of the rivet. The shear on the line A-A is equal to the force P.

Taking the section of the rivet in the upper half of the plate, we shall consider the forces under which this is in equilibrium. This piece may be considered as a column with an eccentric load acting at its edge A-A. The stresses resisting this are then determined by the well known formulas

$$p = \frac{P}{t}\left(1 + \frac{6e}{t}\right) \text{ and } p' = \frac{P}{t}\left(1 - \frac{6e}{t}\right)$$

With e equal to $\frac{t}{2}$, $p = \frac{4P}{t}$ and $p' = -\frac{2P}{t}$.

This would indicate that there is a tension p' at the outside edge between the rivet and the plate, a situation which is impossible.

## STRESSES IN STRUCTURAL RIVETS

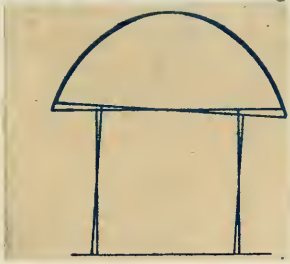
Inasmuch as the rivet fits tightly in the hole, a pressure will ensue on the opposite side of the rivet as soon as the rivet tends to leave the metal on the right side of the hole. Then $p'$ is a pressure on the outer edge of the plate at the left hand side of the hole.

If pressures in one direction are considered positive and in the other direction negative, the net sum of the pressures is $\frac{pt}{4} = P$. The moment of these pressures is $\frac{pt^2}{8} = \frac{Pt}{2}$. Therefore there must be a moment in the rivet section A-A equal to $\frac{Pt}{2}$. The half of the rivet in the other plate being under analagous but opposite forces, the whole is in equilibrium. Each plate has a net pressure equal to P which resists the load, and is further acted upon by a moment $\frac{Pt}{2}$, evidences of which are so often apparent in a lap joint.

The stresses on the rivet shank then may be resolved into those due to bending moment and those due to shear, and these for convenience may be considered separately. The moment on the rivet tends to rotate it until the reacting pressures balance the moment. We have considered the reactive stresses on the shank neglecting the effect of the rivet head. The uniform pressure on the

shank due to shear does not affect the head except to slide it to a slight extent on the surface of the plate. The moment, however, brings one edge of the head to bear upon the plate and the other edge to leave the plate. A hot driven rivet has initial tension in it due to shrinkage, so that any tendency of the rivet head to leave the plate is in reality a relief of the initial pressure under the head, whereas the pressure due to moment is an increase in pressure over the initial pressure. The stresses then due to moment may be analyzed as if the head were integral with the plate as far as forces perpendicular to the surface between is concerned whether it be tension or compression.

Neglecting for the present the stresses due to shear, the problem is to find the distribution of stress under the head and along the shank due to moment.



The rivet under stress will tip thru an angle. The deflection of the head with respect to the shank is represented by the diagram. We will assume that the stress varies as this deflection. If now we can find the pressure at the edge of the head due to a certain

deflection and can also determine the pressures on the shank due to the same deflection, we can derive an expression for the combination. The summation of the moments of these pressures will equal the moment of the shear and determine the maximum pressures in terms of the load.

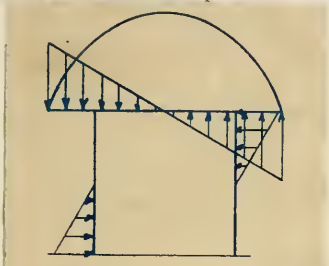


FIGURE 3

The resultant pressures then due to moment are as indicated in figure 3 and the sum of their moments equals the moment at the section or $\frac{Pt}{2}$. The problem now is to find the relative values of the pressures at the edge of the plate on the shank and at the edge of the head.

In the first place the distribution of stress and resultant deflection in the plate and in the shank due to the pressures between the two, is indeterminate by any theoretical consideration. The same may be said regarding the stresses and deflection due to pressures under the head. We therefore must resor t to an experimental determination of these relations.

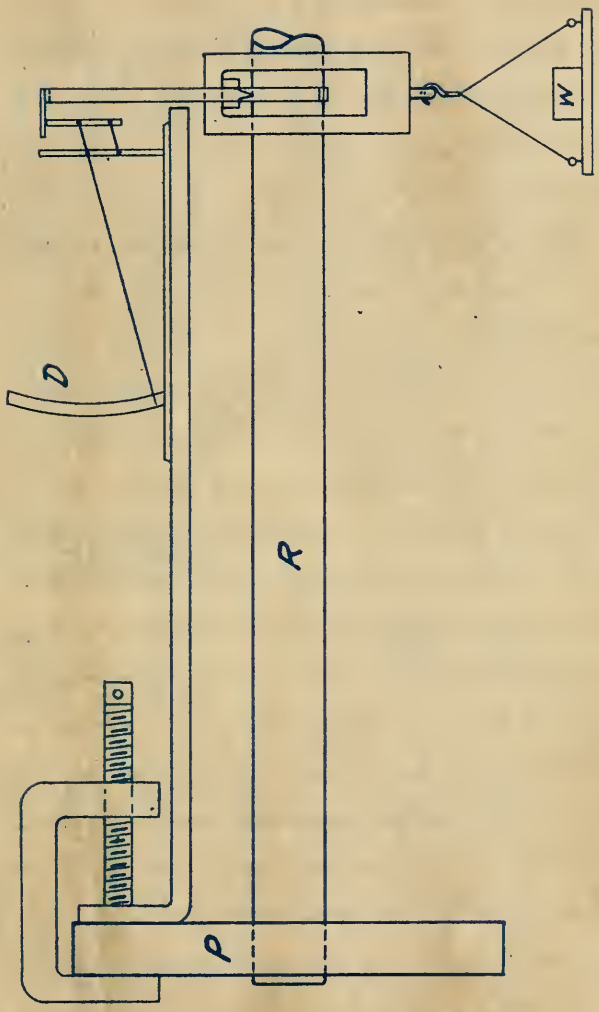



FIGURE 4

Figure 4 is a diagram of the arrangement of apparatus for an experiment to determine the deflection of a rivet shank alone in a plate when the former is subjected to a bending moment. P is a $\frac{3}{4}$" Plate, measuring about 6" X 8". In the center of the plate a one inch diameter hole was drilled to receive the one inch diameter rod with a driving fit. The pan with weight W is hung by a sharp point at a distance of 12" from the center of the plate. A support for the deflectometer is clamped to the plate so that all deflections recorded will be with respect to the plate and any disturbance or twist of the plate will be automatically corrected, i.e. it will not be recorded on the deflectometer. A wire wound around the rod at the point of support of the pan is hooked over the deflectometer to operate the same. Increments of load of 25# each were placed upon the pan and the deflection in each case read. The 25# increment which made an aggregate of 325# gave a greater increment deflection than have been uniformly read for the other 25# loads and the next 25# increment gave a still greater increment of deflection showing that the elastic limit had been reached. In removing the weights it was observed that the deflectometer readings for the different

loads were greater than those for the same loads during the run, showing a permanent set.

The total deflection recorded by the deflectometer is made up of two components. First, there is a deflection due to the bending of the rod. Secondly, there is a deflection due to the slope of the tangent to the neutral axis at the hole resulting from the deflection of shank and plate. The former deflection may be computed by the cantilever formula, the section of the rod being known. This then subtracted from the total deflection gives the deflection due to slope of the tangent. From this slope may be computed the deflection at the edge of the plate. The pressure at the same place calculated from the bending moment on the rod.

The data for this experiment is tabulated in the log sheet in Part IV. A curve for the deflections is shown also.

From the data it is seen that a load of twenty five pounds gives a deflection of 0.01 inch. The formula for cantilever deflection with load at the end is

$$F_M = \frac{Pl^3}{3EI} = \frac{25 \times 12 \times 12 \times 12 \times 64}{3 \times 30000000 \times 3.1416} = .009778"$$

The deflection due to the slope of the tangent equals

$$F_s = .010000 - .009778 = .000222''$$

Considering the respective lever arms the deflection at the edge of the plate is

$$F_p = .000222 \times \frac{3/8}{12} = .000006945''$$

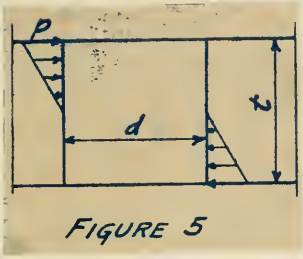


FIGURE 5

Any set of pressures resisting the moment in a rivet shank are represented by the diagram. If we call the average pressure across the rivet at the edge of the plate P and the thickness of the plate t, then the resisting moment equals $\frac{pdt}{2X2} \times \frac{2}{3}t = \frac{pdt^2}{6}$.

Placing the resisting moment equal to the imposed moment $\frac{pdt^2}{6} = 300$. In thise case d = 1" and t = $\frac{3}{4}$", hence

$$p = \frac{1800}{1X9/16} = 3200 \text{ lbs. per sq. in.}$$

The deflection per 1000 lbs. per sq. in. pressure equals

$$F_s = \frac{.000006945}{3.2} = .000002102''$$

We have therefore experimentally determined the relation between deflection and resistance of the shank to bending moment. It still remains to find the same relations in the head of the rivet.

# PART II

A consideration of the resistance of the rivet head to bending moment.

## RESISTANCE OF HEAD TO BENDING MOMENT

The resistance of the rivet head to bending moment in the rivet is due largely to the initial tension in the rivet. In order to have this experiment conform as nearly as possible with practical circumstance, an initial pressure should be induced between the head and the plate. This should approximate as nearly as possible that due to rivet shrinkage.

The stress due to shrinkage is a very doubtful quantity. A purely theoretical consideration gives us some very awkward results. In order that we may have some ideas upon which to base assumptions, we shall run thru a theoretical consideration of the matter.

When a rivet is driven, the plates are pressed together with considerable force by the machine. When the dies are released, the rivet is usually about of a cherry red heat, sometimes cooler sometimes warmer. As an average and for purposes of definiteness we shall call the temperature upon release that of cherry red which is 900° C according to Merriman's "American Civil Engineer's Hand Book" page 1246. Now we can safely say that the pressure exerted by the machine exceeds by far the strength of the rivet at this temperature.

From Kent's "Mechanical Engineer's Hand Book" page 464 we learn that steel at 2000° F has 5% of the strength it has at 0°, at 1800°F, 7.5%; at 1600°F, 10%. Interpolating in the table and transposing to Centigrade scale we find the strength for the Rivet at 900°C equals 9.3% of the strength at 0° or

$\frac{9.3}{100}$ X 50000 4650 lbs. per sq. in.

The above then is the tension put in the rivet by the machine.

From Kent's Handbook, page 567 we find the following coefficients of expansion at high temperature for steel whose composition approximates that of rivet steel

| 1.5° - 200°C | 200° - 500° | 500° - 650° | 800° - 950° |
|---|---|---|---|
| $11.5 \times 10^{-6}$ | $14.5 \times 10^{-6}$ | $17.5 \times 10^{-6}$ | $23.3 \times 10^{-6}$ |

From this table we determine that the average coefficient for the 900° is $16.7 \times 10^{-6}$. If we designate the grip of the rivet by 1, the shortening is equal to $e = 900 \times 16.7 \times 10^{-6} \times 1$

.01502 1

If we consider that the plates are rigid then all the stretch must be in the rivet. This is expressed by the formula

$e = \frac{Pl}{AE}$ or substituting S for $\frac{P}{A}$

$e = \frac{Sl}{E}$. Equating this to the expression for e due to shrinkage.

$\frac{Sl}{E}$ .01502 l

S .01502 E .01502 X 30000000 450600#/0"

We have figured this considering the plates rigid. Now the plates actually compress together and if they had perfect bearing on each other and if the area of the plate affected by the pressure equaled the area of the rivet section, then the l in our formula $e = \frac{Pl}{AE}$ would be twice as large and S would be half as much. The facts are that a much greater area of the plate than that equal to the rivet section is affected, which would tend to keep the unit stress well up. On the other hand there are invariably layers of rust, paint, dirt, etc., between the metal surfaces which compress easily. Any imperfect bearing or matching of surfaces would also give room for compression. These of course would take up the shrinkage compression and reduce the value of S. Most of these however are broken down by the riveting machine. Then again as the hot rivet enters the hole, it heats the metal around it and the shrinkage will then be over a range of less than 900° for after a certain temperature both metal and rivet cool and shrink together.

The question is would this value of S above plus the value for original machine pressure, a total of about 455200 #/□", be reduced 10 or 20 times by the above factors. One

tenth of S is just under the ultimate strength; one twentieth is just under the yield point. It is our opinion that twenty is a high factor in any ordinary case by which to reduce this stress. We have the idea that immediately after driving and during cooling the rivet is stressed up to the yield point and the metal flows as the rivet shrinks till the rivet and the plate reach the same temperature. From then on the two shrink together, each the same amount so that the stress in the rivet remains the same.

In order that we may have something definite to work with, we shall stress the rivet up to one half of the elastic limit or about 15000 lbs. per sq. in. This will allow the stress to vary on one side down to zero and on the other up to the elastic limit. Then the uniform variation of stress will hold thruout.

Figure 5 is a diagram of the apparatus as set up for this experiment. A 1" diam. rod with a Carnegie Standard one inch rivet head was turned out of a piece of cold rolled soft steel, the same material as was used for the previous experiment. This was fitted in a tapered hole in a structural steel plate measuring $\frac{3}{4}$"X8"X 12". The hole was tapered ao as to fit the rod under the head and yet so as to leave the rod free from the sides of the hole. A net section of one inch diameter was maintained for twelve inches and beyond this a one inch eye was forged and welded on the end. A $\frac{3}{4}$" cable drawn thru the eye was fastened with both ends in the jaws

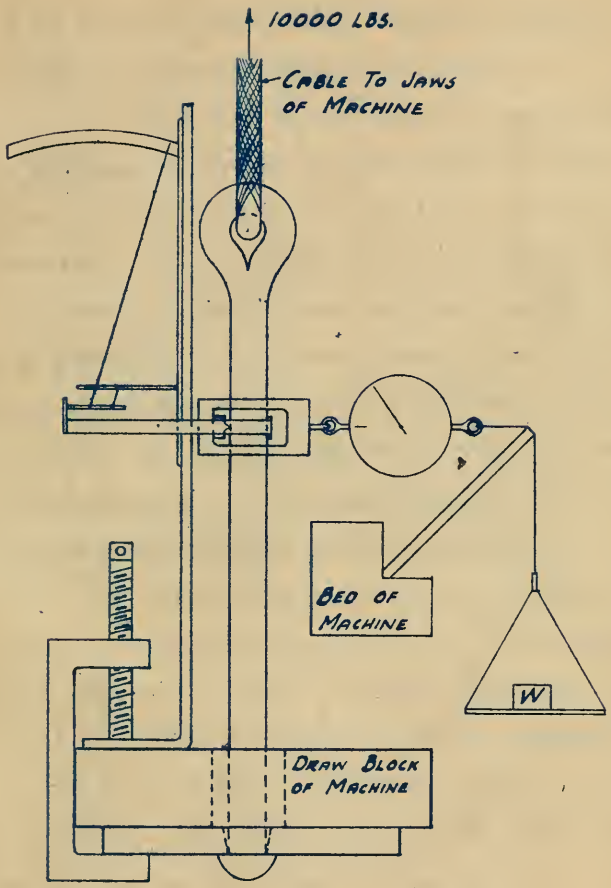

10000 LBS.
CABLE TO JAWS OF MACHINE
BED OF MACHINE
W
DRAW BLOCK OF MACHINE

FIGURE 6

of the testing machine. Then a tension of 10000# was maintained on the rod during the experiment.

At a point twelve inches from the flat face of the head, a horizontal load was applied to the rod thru a ballbearing. The loads were placed on the pan and the horizontal pull read on the dynamometer. The deflections for the different loads are recorded in the log sheets, Part IV. The curve drawn for total horizontal load against deflection shows that there is a deflection of 0.0579" for 300 lbs. load. Part of this load however acts to pull the taut cable out of line. To solve for this we must use several deflection formulas.

The distance from the center of hinge at the eye to the face of the head is 18.625". The distance from the same point to the jaws of the machine is 30". Then if $f_e$ is the deflection of the eye from the straight line between head and jaws, the reaction R at the eye due to the load P, which is that portion of the 300 lbs. used to pull the rod out of line, is equal to

$$\left( \frac{f_e}{30} + \frac{f_e}{18.6} \right) \times 10000 \text{ lbs.}$$

The deflection $f_e$ is made up of several

parts. First there is a deflection due to the distortion of the head and equals

$$\left( 0.0579 - \frac{( 300 - P ) \times 12^3}{3\ EI} \right) \times \frac{18.625}{12}$$

which is the total deflection at P as read on the deflectometer minus the deflection at the same point due to bending as a cantilever from ( 300 - P ), and then this deflection multiplied by the ratio of lever arms to get the deflection at the eye.

The deflection at the eye from the load ( 300 - P ) inducing bending flexure is found from the cantilever formula of deflection for a load at a distance Kl from the free end which is

$$f = \frac{Pl^3}{6\ EI} ( 2 - 3K + K^3 )$$

where $K = \frac{6.625}{18.625}$ and $l = 18.625$ and P is ( 300 - P ).

The deflection at the eye in the opposite direction due to R is $\frac{18.625}{12}$ times that at P due to the load P.

The load P is acting on a beam, fixed at one end and supported at the other, at a distance Kl from

the supported end. The formula for deflection at P is

$$f = \frac{Pl^3}{12\ EI}\ (\ 8K^3 - 3K^2 - 6K^4 + K^6\ )$$

where K is again equal to $\frac{6.625}{18.625}$ and $l = 18.625$. The reaction at the supported end for this kind of beam equals $\frac{1}{2}P(2 - 3K + K^2)$. The algebraic sum of these deflections multiplied by $(\ \frac{1}{30} + \frac{1}{18.625}\ ) \times 10000$ is equal to the reaction:-

$$\left[(0.0579 - \frac{(300 - P) \times 12^3}{3\ EI})\ \frac{18.625}{12} + \frac{(300 - P)\ 1^3}{6\ EI} \times \right.$$

$$\left. (2 - 3K + K^3) - \frac{Pl^3}{12\ EI}\ (8K^3 - 3K^2 - 6K^4 + K^6) \times \frac{18.625}{12}\right] \times$$

$$(\frac{1}{30} + \frac{1}{18.625}) \times 10000 = \tfrac{1}{2}P\ (\ 2 - 3K + K^2\ ).$$

Substituting the proper values for K and l, P = 165 lbs. and ( 300 - P ) = 135 lbs.

The deflection at the load due to bending in the rod is

$$\frac{135 \times 12^3}{3\ EI} = 0.0528"$$

The deflection due to head distortion is equal to 0.0578 - 0.0579 = 0.0051". The deflection under the head at the rivet shank by ratio of lever

arms eqals $\frac{0.0051}{2 \times 12} = 0.000212"$. The stress at the same point equals

$$S = \frac{Mc}{I} = \frac{135 \times 12 \times 32}{3.1416} = 16500^{\#} \text{ per sq. in.}$$

The deflection of the head per $1000^{\#}$ per sq. in. stress equals

$$f_h = \frac{0.000212}{16.5} = 0.0000129"$$

This value is over six times as much as that for the same stress between shank and plate.

If $v_h$ is the deflection of the head at the edge of the shank and $v_s$ is the deflection of the shank at the edge of the plate, then $\frac{v_h}{v_s} = \frac{d}{t}$. $S_h$, the unit stress for deflection $v_h$, equals $\frac{v_h}{f_h}$; and the moment resisted by the head equals

$$\frac{SI}{c} = \frac{3.1416\ d^3 v_h}{32\ f_h}$$

The resistance of the shank to bending moment equals

$$\frac{pdt}{4} \times \frac{2}{3}t = \frac{p\ d\ t^2}{6}$$

where p is the average stress across the shank at the edge of the plate and equal to $\frac{v_s}{f_s}$. The total bending moment then equals

$$\frac{P\ t}{2} = \frac{p\ d\ t^2}{6} + \frac{3.1416}{32} \times \frac{v_h}{f_h} \times d^3$$

$$v_s = p\, f_s \qquad \text{therefore} \qquad v_h = \frac{p\, d\, f_s}{t}$$

$$\frac{Pt}{2} = \frac{p\, d\, t^2}{6} + \frac{3.1416}{32} \times \frac{p\, d^4}{t} \times \frac{f_s}{f_h}$$

From the above

$$p_M = \frac{Pt}{\frac{d\, t^2}{3} + \frac{3.1416\, d^4\, f_s}{16\, t\, f_h}}$$

which is the maximum pressure for moment only.

Added to this is the pressure due to shear and equal to $\frac{P}{td}$.

The total maximum then equals

$$p = \frac{P\, t}{\frac{d\, t^2}{3} + \frac{3.1416\, d^4\, f_s}{16\, t\, f_h}} + \frac{P}{td}$$

In part IV are included some tables giving the maximum bearing stresses for different combinations of plate thickness and rivet diameter.

## PART III

## Variation of Stress Across the Rivet.

## VARIATION OF STRESS ACROSS THE RIVET

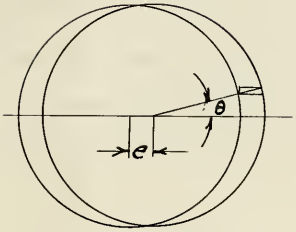



Figure 7

Assume that in Figure 7 the rivet due to stress has had its center moved a distance e. If we assume that there is no friction between the shank and the shank and the plate, the stress at any point is proportional to the displacement of the line of contact and acts normal to the curve. This deflection equals e cos θ, and the normal stress equals e cos θ x E. The component in the direction to resist shear is e $\cos^2$ θ x E. Summing these up across the diameter of the rivet

$$p\,d = 2\,E\,e \int_0^{\frac{1}{2}d} \cos^2 \theta \, dy.$$

$\cos^2 \theta = 1 - \frac{4\,y^2}{d^2}$ and E e = S, the stress at the point of greatest deflection.

$$p\,d = 2\,S\int_0^{\frac{1}{2}d}\left(1 - \frac{4\,y^2}{d^2}\right)dy = 2\,S\left[y - \frac{4\,y^3}{3\,d^2}\right]_0^{\frac{1}{2}d}$$

$$= 2\,S\left(\frac{d}{2} + \frac{d}{6}\right) = \frac{2}{3}\,S\,d$$

Therefore $S = \frac{3}{2}\,p$, or the maximum stress in a lamina across the diameter of a rivet is one and one half times the average. This holds for any circular section rivet whether it be in a lap joint or butt joint.

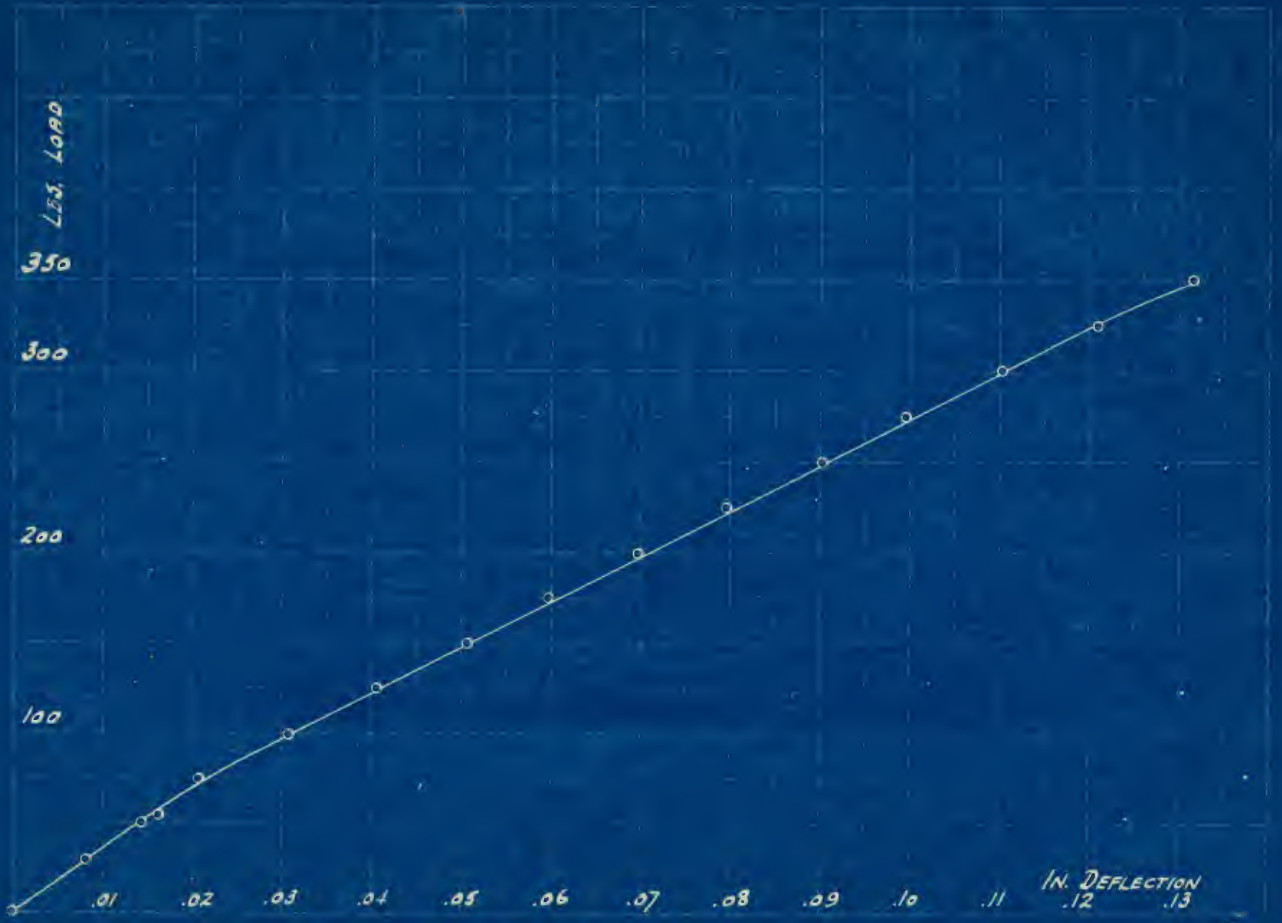
# PART IV

Log Sheets, Curves, and Photographs of Experiments Performed.

## Log of Experiment for Shank Deflection.

Plate -- 8"x 12"x $\frac{5}{8}$". Rod -- 1"Diam. Lever Arm -- 12"

| Load | Deflectometer Reading | Net Deflection |
|---|---|---|
| 0 | 0.0500 | 0.0000 |
| 30 | 0.0580 | 0.0080 |
| 50 | 0.0642 | 0.0142 |
| 55 | 0.0660 | 0.0160 |
| 75 | 0.0704 | 0.0204 |
| 100 | 0.0805 | 0.0305 |
| 125 | 0.0905 | 0.0405 |
| 150 | 0.1005 | 0.0505 |
| 175 | 0.1097 | 0.0597 |
| 200 | 0.1196 | 0.0696 |
| 225 | 0.1295 | 0.0795 |
| 250 | 0.1400 | 0.0900 |
| 275 | 0.1495 | 0.0995 |
| 300 | 0.1600 | 0.1100 |
| 325 | 0.1708 | 0.1208 |
| 350 | 0.1815 | 0.1318 |



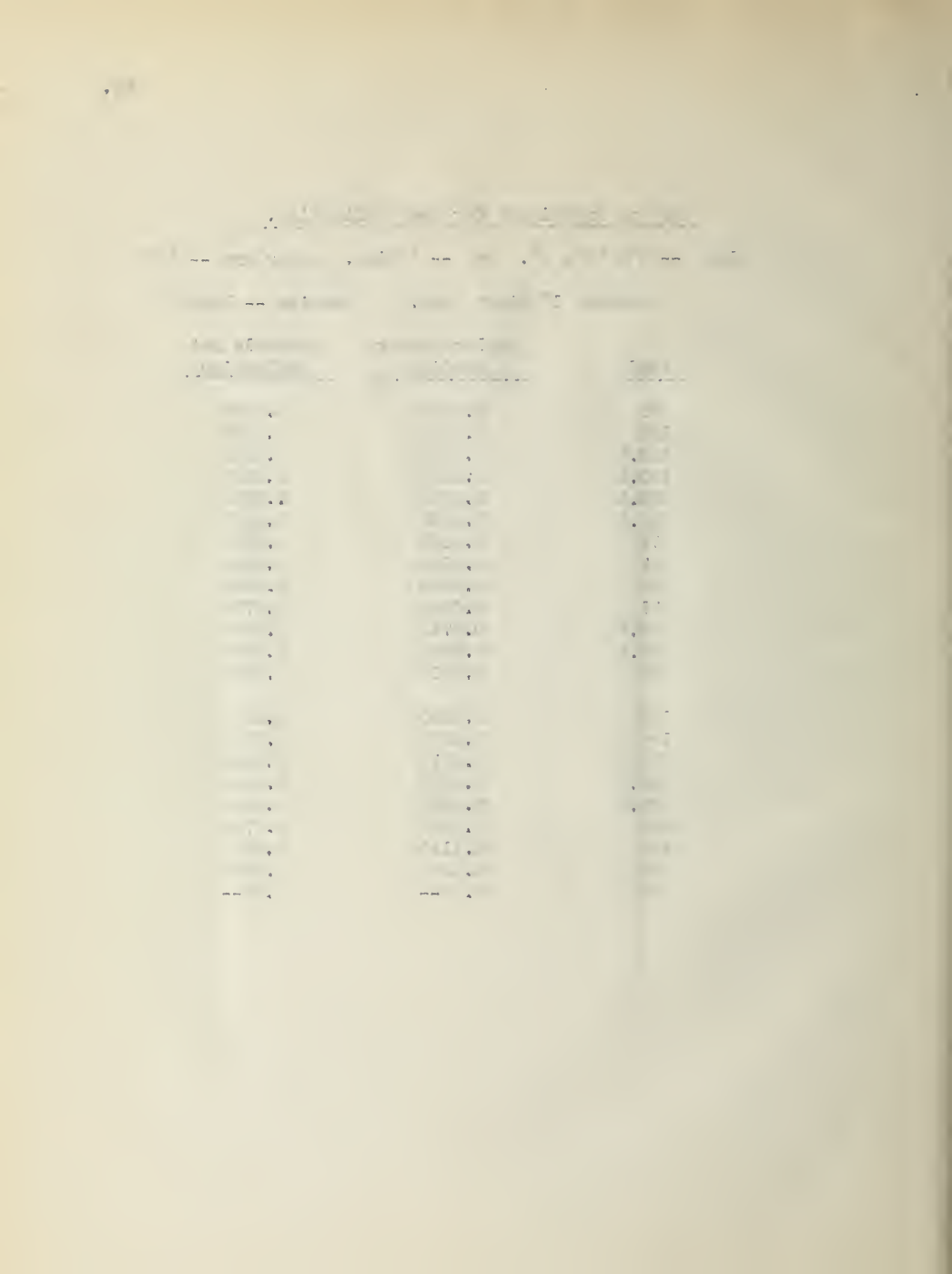

Curve For Shank Resistance Experiment.

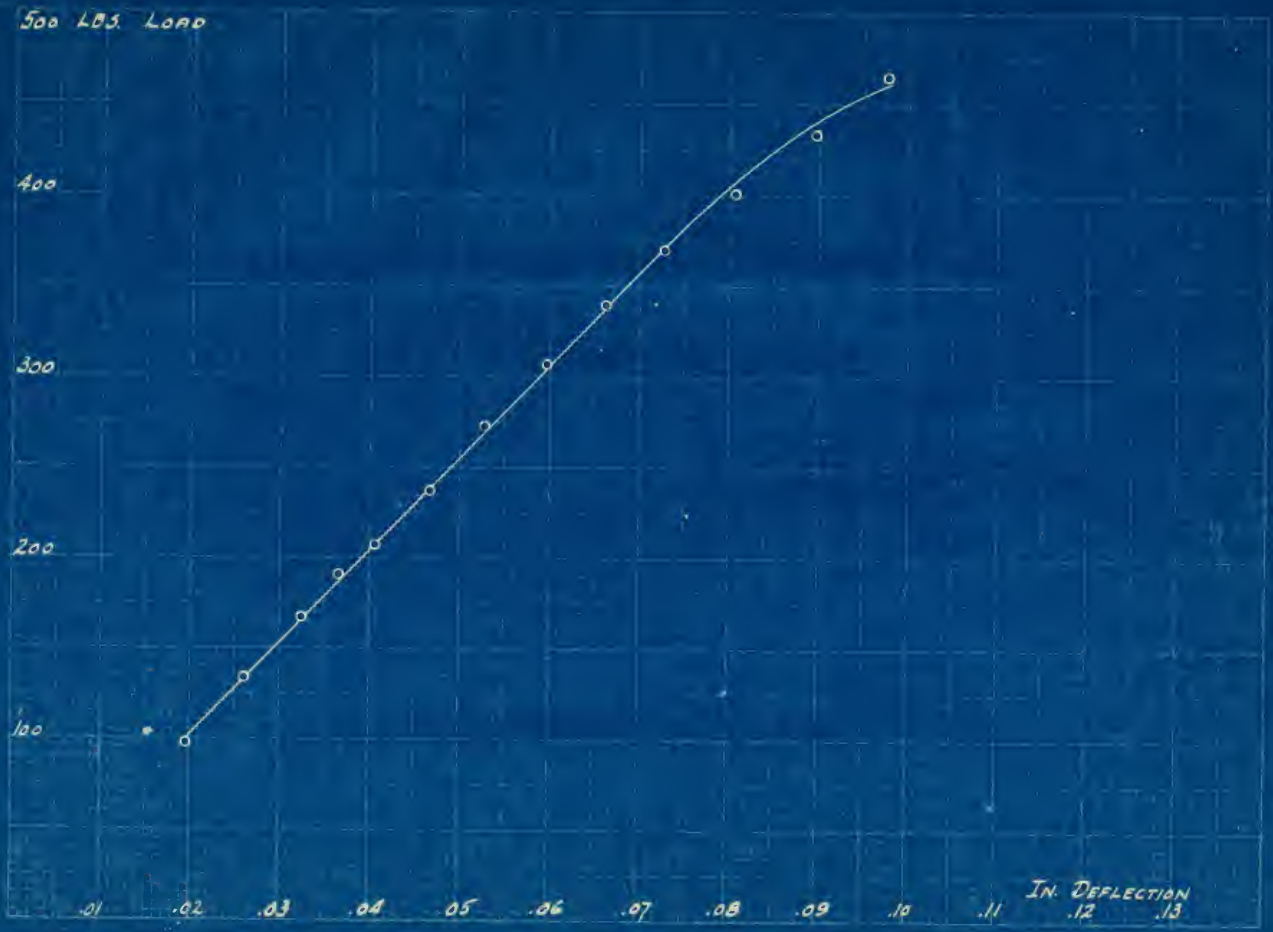
## Log of Experiment for Head Deflection.

Plate -- 8"x 12"x $\frac{3}{4}$". Rod -- 1"Diam. Lever Arm -- 12"

Standard 1" Rivet Head. Tension -- 10000#

| Load | Deflectometer Reading | Probable Net Deflection |
|---|---|---|
| 99 | 0.0138 | 0.0198 |
| 135 | 0.0200 | 0.0260 |
| 166.5 | 0.0262 | 0.0322 |
| 190.5 | 0.0306 | 0.0366 |
| 206.5 | 0.0346 | 0.0406 |
| 238.5 | 0.0405 | 0.0465 |
| 274 | 0.0466 | 0.0526 |
| 307 | 0.0536 | 0.0596 |
| 340 | 0.0600 | 0.0660 |
| 371 | 0.0666 | 0.0726 |
| 402.5 | 0.0745 | 0.0805 |
| 434.5 | 0.0835 | 0.0895 |
| 465 | 0.0917 | 0.0977 |
| | | |
| 143 | 0.0530 | 0.0250 |
| 194 | 0.0620 | 0.0340 |
| 250 | 0.0718 | 0.0438 |
| 313.5 | 0.0828 | 0.0548 |
| 367.5 | 0.0928 | 0.0648 |
| 424 | 0.1045 | 0.0765 |
| 456 | 0.1110 | 0.0830 |
| 493 | 0.1200 | 0.0920 |
| 526 | 0.13-- | 0.10-- |






CURVE FOR HEAD RESISTANCE EXPERIMENT.

Values of $p_M$, maximum average stress due to bending moment, in terms of $\frac{P}{t\,d}$, for different combinations of plate thickness and rivet diameter.

| Plate Thickness | Rivet Diameter $\frac{5"}{8}$ | $\frac{3"}{4}$ | $\frac{7"}{8}$ | 1" |
|---|---|---|---|---|
| $\frac{5"}{16}$ | 1.7 | 1.29 | 0.97 | 0.73 |
| $\frac{3"}{8}$ | 2.08 | 1.70 | 1.35 | 1.06 |
| $\frac{7"}{16}$ | 2.34 | 2.02 | 1.70 | 1.40 |
| $\frac{1"}{2}$ | 2.53 | 2.26 | 1.98 | 1.70 |
| $\frac{9"}{16}$ | 2.65 | 2.45 | 2.20 | 1.95 |
| $\frac{5"}{8}$ | 2.74 | 2.50 | 2.37 | 2.16 |
| $\frac{11"}{16}$ | 2.80 | 2.67 | 2.50 | 2.32 |
| $\frac{3"}{4}$ | 2.84 | 2.74 | 2.60 | 2.45 |
| $\frac{13"}{16}$ | 2.87 | 2.79 | 2.68 | 2.55 |
| $\frac{7"}{8}$ | 2.90 | 2.83 | 2.74 | 2.63 |
| $\frac{15"}{16}$ | 2.92 | 2.86 | 2.78 | 2.69 |
| 1" | 2.94 | 2.88 | 2.82 | 2.74 |

Values of P, maximum average stress due to bending moment and shear, in terms of $\frac{P}{td}$, for different combinations of plate thickness and rivet diameter

| Plate Thickness | RIVET DIAMETER 5/8" | 3/4" | 7/8" | 1" |
|---|---|---|---|---|
| 5/16" | 2.7 | 2.29 | 1.97 | 1.73 |
| 3/8" | 3.08 | 2.70 | 2.35 | 2.06 |
| 7/16" | 3.34 | 3.02 | 2.70 | 2.40 |
| 1/2" | 3.53 | 3.26 | 2.98 | 2.70 |
| 9/16" | 3.65 | 3.45 | 3.20 | 2.95 |
| 5/8" | 3.74 | 3.50 | 3.37 | 3.16 |
| 11/16" | 3.80 | 3.67 | 3.50 | 3.32 |
| 3/4" | 3.84 | 3.74 | 3.60 | 3.45 |
| 13/16" | 3.87 | 3.79 | 3.68 | 3.55 |
| 7/8" | 3.90 | 3.83 | 3.74 | 3.63 |
| 15/16" | 3.92 | 3.86 | 3.78 | 3.69 |
| 1" | 3.94 | 3.88 | 3.82 | 3.74 |

Values of S, maximum stress due to bending moment and shear, in terms of $\frac{P}{td}$, for different combinations of plate thickness and rivet diameter

| Plate Thickness | RIVET DIAMETER 5/8" | 3/4" | 7/8" | 1" |
|---|---|---|---|---|
| 5/16" | 4.05 | 3.44 | 2.94 | 2.59 |
| 3/8" | 4.62 | 4.05 | 3.53 | 3.09 |
| 7/16" | 5.01 | 4.53 | 4.05 | 3.60 |
| 1/2" | 5.30 | 4.89 | 4.47 | 4.05 |
| 9/16" | 5.48 | 5.18 | 4.80 | 4.43 |
| 5/8" | 5.61 | 5.25 | 5.05 | 4.74 |
| 11/16" | 5.70 | 5.51 | 5.25 | 4.98 |
| 3/4" | 5.76 | 5.61 | 5.40 | 5.18 |
| 13/16" | 5.81 | 5.69 | 5.52 | 5.33 |
| 7/8" | 5.85 | 5.75 | 5.61 | 5.45 |
| 15/16" | 5.88 | 5.79 | 5.67 | 5.54 |
| 1" | 5.91 | 5.82 | 5.73 | 5.61 |

## CONCLUSION

From the tables we notice that the maximum bearing stress on a lap joint rivet may be as much as 5.91 times that usually figured in practice. This has been figured on the assumption that the rivet head takes the maximum bending moment possible. Furthermore we have assumed that the rivet completely fills the hole and is in contact with the metal at every point. There is every reason to doubt that neither of these ideal conditions obtain in practice. If the head is so driven that it takes no bending moment, then the maximum average pressure across the shank at the edge of the plate is four times that commonly figured. The absolute maximum is six times that figured. In cases where the rivet does not completely fill the hole, this ratio may run much higher.

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